N62-16292

NASA TN D-1404

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42p

TECHNICAL NOTE

D-1404

EFFECT OF NINE LUBRICANTS ON ROLLING-CONTACT FATIGUE LIFE

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

WASHINGTON October 1962

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SUMMARY

The NASA fatigue spin rig and five-ball fatigue tester were used to determine the rolling-contact fatigue lives at room temperature and at 300° F of groups of AISI M-2 and M-1 alloy steel balls run with nine lubricants having varied chemical and physical characteristics. These lubricants were classified as three basic types: esters, mineral oils, and silicones. A correlation is suggested between the plastically deformed-profile radius of the test-specimen running track at ambient temperature after 30,000 stress cycles of operation and rolling-contact fatigue life.

The fatigue results obtained in this investigation did not correlate with temperatures measured at the edge of the contact zone of the test specimens during rolling contact in the five-ball fatigue tester. The lubricants giving the longest lives generally yielded the highest measured temperatures. However, at an outer-race temperature of 300° F, the measured contact temperature differences among the lubricants were insignificant. Longer fatigue lives were obtained at room temperature with the silicone and the dioctyl sebacates than with other lubricants; at 300° F, however, the silicone and the mineral oils gave the longer lives. The relative order of fatigue results obtained in the five-ball fatigue tester compared favorably with that obtained with 7208-size bearings tested with the same lubricants.

INTRODUCTION

The precise role of a lubricant in rolling-contact fatigue is not fully understood. In past years, for most purposes it was only necessary to specify a particular grade of mineral oil for the lubrication of a rolling-element bearing. However, as technology advances, higher bearing temperatures (i.e., in the range 500° to 800° F) dictate the need to develop new lubricants with better thermal and oxidative stability. Knowledge of how the chemical and physical properties of the lubricant affect rolling-contact fatigue life would be useful as a guide in developing new lubricant formulations.

Differences in the properties of lubricants can affect at least two variables during rolling contact. They are contact temperature (refs. 1 and 2) and contact geometry (refs. 3 and 4). The heat generated and the resulting temperature in the zone of contact of two rolling surfaces are functions of the rheological and thermal properties of the lubricant; therefore, contact temperature may vary with different lubricants under dynamic conditions. The generation of heat within the lubricant film induces thermal gradients and thus thermal stresses in the material, which in turn can affect the shearing stresses in the material as calculated from theory (ref. 5). Since it is generally accepted that fatigue life varies inversely with stress to the ninth power, this increase in the shearing stresses may account for differences in fatigue lives exhibited with different lubricants. Changes in contact area may produce differences in contact stress. The differences in contact stress with various lubricants may, then, be a second factor that accounts for differences in fatigue lives with different lubricants.

The research reported herein was undertaken to determine the effect of changes in various operating variables induced by lubricant type on bearing fatigue and load capacity. The objectives were the following:

(1) To investigate the effect of lubricant type or base stock on rolling-contact fatigue life;

(2) To determine the effect of lubricant type on contact geometry;

(3) To determine experimentally a relation between contact temperature, lubricant type, and fatigue life.

Tests were run at ambient temperature with nine lubricants having varied chemical and physical characteristics and 9/16-inch AISI M-2 steel balls at a maximum Hertz stress of 725,000 psi in the fatigue spin rig. Four of these same lubricants were tested with 1/2-inch M-1 steel balls in the five-ball fatigue tester at a maximum Hertz stress of 800,000 psi and a 300° F race temperature. Fatigue-life results were evaluated with respect to the plastic deformation and wear and the temperature near the contact zone. A comparison of these data was also made with full-scale bearing fatigue results obtained elsewhere for 7208-size angular-contact bearings. All the experimental results for a given lubricant were obtained from the same lubricant batch, including those for the full-scale bearings.

BACKGROUND

There is much data in the literature on the effect of the various physical and chemical properties of lubricants on rolling-contact fatigue. Physical properties of a lubricant include its thermal and rheological properties. The term thermal properties is self-explanatory; rheological properties define the flow and the deformation characteristics of a fluid. For a Newtonian fluid the flow behavior is specified by a single rheological property, the fluid viscosity. The variation of viscosity with temperature and pressure, therefore, serves to define the flow behavior of the fluid completely. Certain fluids, including some lubricants, exhibit non-Newtonian behavior of the pseudoplastic and/or elastic type. Pseudoplastic behavior is characterized by a shear stress dependent on fluid viscosity; that is, the viscosity decreases as the shear rate increases. Elastic behavior is characterized by the viscosity being dependent upon the previous rheological history of the fluid; that is, the fluid viscosity is a function of the magnitude and the duration of the imposed stress (ref. 6). Words such as "relaxation" and "fluid memory" are often used in describing elastic behavior.

Investigators have shown that, as the viscosity at atmospheric pressure of a particular lubricant type is increased, the life of a bearing element also increases (refs. 7 to 11). If the lubricant pressure-viscosity coefficient is increased by changing the lubricant base stock, longer fatigue life is obtained for a given lubricant viscosity at atmospheric conditions (refs. 8 and 9). Therefore, it has become generally accepted that increased life is obtained with increased viscosity and higher pressure-viscosity coefficients. For some lubricant types, however, fatigue life either remains unchanged or decreases with increasing bulk lubricant viscosity and pressure-viscosity coefficient (refs. 10 to 13).

Generally, the introduction of additives into lubricants has produced little effect on fatigue life. When used in substantial percentages, however, viscosity improver type additives, such as methacrylate, can change the rheological properties of a lubricant even to the extent of changing a Newtonian fluid into a non-Newtonian one (refs. 14 and 15). The lubricant can then act in the manner of a pseudoplastic or elastic fluid. Reference 7 reports that the addition of such an additive to a dibasic acid ester produced no significant effect on the fatigue life of full-scale bearings. Reference 16 reports that a series of extreme-pressure additives added to a paraffinic mineral oil did not affect fatigue life. There is no evidence showing that extreme-pressure additives affect rheological properties.

It has been suggested (ref. 12) that fatigue life is affected by the number of aromatic or naphthenic rings in the lubricant molecule.

Reference 12 also suggests that, if the polarity or reactivity of the molecule is increased, decreased fatigue life can be expected.

All the aforementioned variables can be important to the rolling-contact fatigue phenomenon.

APPARATUS

The rolling-contact fatigue spin rig (figs. 1(a) and (b)) and the five-ball fatigue tester (figs. 1(c) and (d)) were used in these tests.

Fatigue Spin Rig

A detailed description of the NASA fatigue spin rig can be found in reference 17. Figure 1(a) is a cutaway view of this rig. Essentially the fatigue spin rig consists of two balls revolving in a horizontal plane on the bore of a race cylinder (see fig. 1(b)). (The size of the race cylinder is approximately the size of the outer race of a 100 mm aircraft bearing.) Specimen drive is provided by an air jet, and loading is applied by centrifugal force. Instrumentation provides for speed control and automatic failure detection and shutdown.

Five-Ball Fatigue Tester

The NASA five-ball fatigue tester is described in detail in reference 18. Figure 1(c) is a cutaway drawing of this tester. Essentially it consists of a test specimen pyramided upon four lower support balls, positioned by a separator, and free to rotate in an angular-contact raceway (fig. 1(d)). Specimen loading and drive is applied through a vertical shaft. For every revolution of the drive shaft, the test specimen receives three stress cycles. By changing the pitch diameter of the support balls, the contact angle β can be controlled. As in the spin rig, instrumentation provides for automatic failure detection and shutdown.

Operating-Temperature Measuring Device

The five-ball fatigue tester was modified in order to measure the surface temperature near the contact area of a modified test specimen during operation. Figure 1(e) shows the test specimen and the mounting assembly, which is inserted into the drive spindle of the five-ball rig (see fig. 1(c)). Each test specimen had a thermocouple attached with the tip at one edge of the running track. An axial hole was drilled through the drive spindle to accept the thermocouple wire. The thermocouple emf was taken out through a slip-ring brush assembly mounted at the top end of the drive spindle.

TEST LUBRICANTS

The Aeronautical Systems Division of the United States Air Force Systems Command provided the nine experimental lubricants. Each lubricant varied either with respect to base stock, viscosity at atmospheric conditions, or additives. Each was considered of practical interest, and therefore, was selected for fatigue testing in the NASA fatigue rigs previously described. These nine lubricants can be classified as three basic types: esters, mineral oils, and silicones. A summary of the properties of these lubricants can be found in table I. Additional properties can be found in reference 19. Figure 2 is an ASTM standard viscosity-temperature chart for these lubricants.

PROCEDURE

Fatigue Testing

Before assembly in the fatigue spin rig all race cylinders were subjected to a surface-finish and hardness inspection. All test-section components were flushed and scrubbed with ethyl alcohol and wiped dry with clean cheesecloth. Before testing, specimens were weighed, measured, and examined for imperfections at a magnification of 60; at this time, the specimens and all contacting surfaces were also coated with test lubricant. After a test in the fatigue spin rig, the cylinder race track was examined visually for imperfections. If any imperfections were discovered, the test was resumed on a new track of the same cylinder.

Testing procedure for the five-ball fatigue tester was similar to that for the spin rig. The support balls were grouped in sets of four having diameters matched within 20 microinches to ensure even loading of the test specimen at all four contact points. A new set of support balls was used with each test specimen. In both rigs, the speed and oil flow were monitored and recorded at regular intervals.

The stress developed in the contact area under a particular load for both rigs was calculated by using the Hertz formulas given in reference 20.

Method of Presenting Fatigue Results

Total running time for each specimen was recorded and converted into total stress cycles. Failure data were plotted on Weibull coordinates to obtain a plot of the log-log of the reciprocal of the probability of survival as a function of the log of stress cycles to failure. For convenience, the ordinate is graduated in statistical percent of specimens failed. The statistical methods for treating rolling-contact fatigue data given in reference 21 were used to obtain these plots. From these plots the number of stress cycles necessary to fail any given portion of a specimen group may be determined.

Where high reliability is of paramount importance, the main interest is in early failures. For comparison purposes the significant life on the Weibull plot is, therefore, the 10-percent life. The 10-percent life is the number of stress cycles within which one-tenth of the specimens can be expected to fail; this 10-percent life is equivalent to a 90-percent probability of survival. The failure index given with each plot indicates the number of specimens failed out of those tested.

Measurement of Deformation and Wear

Profile traces (fig. 3) of the test specimens run in the five-ball fatigue tester were made on a contour tracer. By use of the contour tracer, the effect of test lubricant on permanent deformation and wear caused by rolling contact was studied.

Deformation and wear data were obtained for M-50 steel balls of a Rockwell C-62 hardness; they were tested at an initial maximum Hertz stress of 800,000 psi and a contact angle of 10° with no heat added for 30,000 stress cycles in the five-ball fatigue tester with each of the nine test lubricants. For each lubricant, eight test-specimen balls were run. Six profile traces were made of each ball at different locations on the surface, perpendicular to the ball running track.

The areas of deformation and wear on a surface trace cannot be measured directly with any degree of accuracy because of their small size on the trace. In order to measure these areas accurately, the surface trace was projected at a magnification of 5 and drawn on polar grid paper. After projection, radial distances of the trace were magnified 50,000 times while circumferential distances were magnified 30 times. The surface trace would then look as shown schematically in figure 3(b). The areas of deformation and of deformation plus wear were measured by use of a planimeter. The wear area is the difference between these two areas. The depth of the track H was also measured.

RESULTS AND DISCUSSION

Rolling-Contact Fatigue

Groups of 9/16-inch vacuum melt AISI M-2 balls of Rockwell C-66 hardness were run in the NASA fatigue spin rig with each of the nine lubricants previously described. Standard test conditions in the spin rig were ambient temperature (i.e., no heat was added) and a maximum Hertz stress of 725,000 psi. The fatigue appearance obtained with specimens run with each of the lubricants was similar. Results of these tests for each of the nine lubricants are given in figure 4 and are summarized in table II.

The Material Laboratory of the New York Naval Shipyard fatigue tested 7208-size thrust ball bearings at 300° F with a 4600-pound thrust load (ref. 22) for six of these nine lubricants. Each of the lubricants was from the same batch as those used in the bench tests reported herein. The data obtained with these six lubricants are replotted in figure 5 by the method of reference 21 and are summarized in table II.

Fatigue tests were conducted in the five-ball fatigue tester with 1/2-inch M-1 steel balls of a Rockwell C-63 hardness with each of four of the six lubricants tested with the bearings and the spin rig. Standard test conditions for these tests were a temperature of 300° F, a drive-shaft speed of 10,000 rpm and a maximum Hertz stress of 800,000 psi. This stress level was chosen because it produces fatigue failures in a reasonable period of time that are the same as those obtained in full-scale bearings and in the spin rig. These data are given in figure 6. A summary of the fatigue lives is given in table II.

The confidence that can be placed in the experimental fatigue results was determined statistically by methods from reference 21. Each of the lubricants tested was compared with NA-XL-7, and confidence numbers for the 10-percent life (table III) were calculated. These confidence numbers indicate the percentage of the time that the 10-percent life obtained with each lubricant will have the same relation to the 10-percent life with NA-XL-7. Thus, a confidence number of 90 percent means that 90 out of 100 times the specimens tested with a given lubricant will give a result similar to those presented in table II.

Load Capacity

An important criterion of bearing operation is its load-carrying capacity, usually termed simply capacity. This is the load that will theoretically give a ball-specimen or bearing life of 1 million stress cycles or bearing inner-race revolutions with a 90-percent probability of survival. The load-carrying capacity of each lubricant tested may be calculated from the fatigue-life results summarized in table II and the equation

$$C = P \sqrt[3]{L}$$

where C is the load capacity; P is the load on the bearing, ball, or test system; and L is the 10-percent life of the bearing or ball specimen. (All symbols are defined in appendix A.) The load capacity obtained with each lubricant tested is given in table IV. It should be noted that the fatigue life and the load-capacity data for each of these lubricants are compared relative to NA-XL-7.

A comparison is made in figure 7 between the relative capacity obtained with the lubricants run in the five-ball rig and the full-scale bearings. A general agreement in the relative order of fatigue lives exists between results for the five-ball rig and the bearings. For the spin rig a comparison of this type with the bearings is not valid because of differences in test conditions (i.e., test temperature, shear rate, and lack of sliding in the spin rig). It is noted from table II, however, that the spin rig ranked the lubricants in somewhat the same order.

Effect of Viscosity on Fatigue Life

In references 8 and 11 it is reported that life varied as the viscosity raised to powers ranging from 0.2 to 0.3. The results obtained with the mineral oils in the fatigue spin rig at room temperature indicate a greater variation of life with viscosity. The results obtained in the five-ball fatigue tester at 300° F, however. agree with those of references 8 and 11. Further investigation along these lines may be warranted.

Effect of Base Stock on Fatigue Life

In order to compare the effects of lubricant base stock on rolling-contact fatigue life, it was deemed advantageous to determine the relative fatigue life with each lubricant at a common viscosity. Using 0.25 as an average exponent in the life-viscosity relation, an adjusted life based on the viscosity of NA-XL-7 was calculated for each lubricant at the same test conditions. These data compared relative to the fatigue lives of NA-XL-7 are presented in table V. If differences in fatigue life of various lubricants are independent of base stock and dependent only on lubricant bulk viscosity, then the adjusted relative lives in table V should all be approximately 1. The values of the relative lives are, however, not unity.

With the exception of the dioctyl sebacate NA-XL-8 at room temperature, the mineral oils generally produced better fatigue lives than the esters under all test conditions. The results for the chlorinated methyl-phenyl silicone lubricant (NA-XL-9) correlated well with the research reported in references 7 to 9, where silicone base stock lubricants exhibited approximately two to six times the fatigue life of mineral oil lubricants of nearly equal viscosity. While the type of silicone lubricant in reference 7 was not specified, a methyl-phenyl silicone was used in the investigations of references 8 and 9. It can, therefore, be concluded from these data and reference information that methyl-phenyl silicone lubricants will generally give longer fatigue lives and thus higher load capacities than ester and mineral oil lubricants of equal or higher viscosities. At room temperature the dioctyl sebacates can give equal or higher lives than the mineral oils of equal or higher viscosity. For the silicone lubricant this result is believed due in part to its greater increase in viscosity with pressure. For the dioctyl sebacates, however, probable pseudoplastic or elastic behavior of these fluids and their surface-adsorbed films may have the greater influence on fatigue life, since these lubricants had lower pressure-viscosity coefficients than the mineral oils (ref. 23).

The results for these lubricants indicate that the bulk viscosity and the pressure-viscosity coefficient are not sufficient criteria by which to gage the effect of a lubricant upon fatigue life. Other factors, such as shear rate (refs. 3 and 4) and relaxation or fluid memory (ref. 24), together with the thermal properties of the lubricant must be considered. These physical properties together with certain chemical factors may affect the contact temperature and the contact stress. Both effects can have a substantial relation to fatigue-life differences between lubricants.

Deformation and Wear

Deformation produced on surfaces in rolling contact can reduce contact stresses. It assumes three basic forms: (1) elastic deformation, (2) plastic deformation, and (3) wear. The latter two forms result in permanent alteration of the ball surface contour, which can be measured after testing. Figure 3(a) is a schematic diagram of the transverse section of a ball surface showing this permanent alteration. A diagram of the surface trace of this transverse section that magnifies the deviation from the true sphere is shown in figure 3(b).

The profile (fig. 3) shows that material has been displaced to the regions on either side of the running track. The region of increased volume extends approximately 1 track width on either side of the contact zone. If wear does not take place, the volume of material displaced to the regions adjacent to the track should equal the volume of material displaced from the track itself, since steels are essentially incompressible. This is not the case, however. The volume lost is much greater than the volume gained; the difference represents the material removed by wear.

Plastic deformation and wear data were obtained for M-50 steel balls of a Rockwell C-63 hardness at an initial maximum Hertz stress of 800,000 psi, a contact angle of 10°, room temperature, and 30,000 stress cycles in the five-ball fatigue tester with each of the nine test lubricants. Average values for the plastic deformation and wear areas are given in table VI.

By the use of trigonometric relations, an effective ball radius r after plastic deformation and wear (which corresponds to the profile radius in fig. 3(a)) at the point of contact can be calculated (see appendix B) in terms of a, H, and R as follows:

$$r = \frac{(a/H)^2}{2\left[R - \sqrt{R^2 - (a/H)^2} - H\right]}$$

On the basis of this equation for r, an effective maximum Hertz stress after 30,000 stress cycles of operation was calculated for each lubricant. These values are also given in table VI. Previous experience has shown that nearly all of the plastic deformation is obtained within 30,000 stress cycles. Table VI indicates that there are differences in the recalculated static stress for specimens run with each lubricant. Theoretical differences in fatigue lives based on these recalculated stresses indicated that stress differences cannot wholly account for the observed differences in life. It must be concluded, however, that, where plastic deformation occurs, calculations of the initial Hertz stress are meaningless.

A plot was made of the 10-percent fatigue life with each lubricant tested in the five-ball fatigue tester as a function of the deformed profile radius (fig. 8). From this figure it is noted that increasing life is obtained with increased profile radius (deformation). Also, ball specimens run with the synthetics, (viz., NA-XL-3 and NA-XL-8) show less plastic flow of the ball material than those run with the mineral oils. While these data are limited, they do suggest that the forces in the lubricant film may distort the stress pattern calculated from the static force conditions to an extent great enough to account for the observed differences in fatigue life with each lubricant. The mechanism through which such a phenomenon can occur cannot presently be explained.

Temperature Studies

In order to determine the variation in contact temperature with lubricant type, temperature measurements were taken at the edge of the running track under different operating conditions with each of the nine test lubricants in the modified five-ball fatigue tester described previously (see fig. 1(e)). Temperature measurements were taken with each lubricant at an initial maximum Hertz stress of 725,000 psi, contact angles of 10° and 30°, and ambient temperature. Measurements were also made with NA-XL-3, NA-XL-4, NA-XL-7, and NA-XL-8 at an initial maximum Hertz stress of 800,000 psi and a 30° contact angle at ambient temperature and at a 300° F race temperature.

The temperature data obtained without heat addition for these lubricants are given in table VII. These temperatures represent a better approximation of the actual temperature in the contact zone of a ball specimen than temperatures measured on the surface of a fixed or nonrotating race.

The lowest temperatures recorded for each operating condition were obtained with the ester group. The silicone lubricant gave the highest temperatures for each of its operating conditions. Temperatures slightly lower than the silicone temperature were recorded for the mineral oil group, while the esters as a group were generally 40° to 60° F lower than both the silicone and the mineral oils.

It has been previously suggested that thermal stresses affect fatigue life to the extent that lubricants yielding the lowest temperature in the contact zone will give the longest fatigue life in rolling contact (ref. 1). When the temperature measurements (table VII) are compared with rolling-contact fatigue-life results (table II), however, it appears that the lubricants giving the longest lives (i.e., the silicone and the mineral oils) yielded the highest measured temperatures. These temperatures were measured with the system running without heat addition. At an outer-race temperature of 300° F, little difference was found in the measured temperatures at the edge of the contact zone despite wide variations in fatigue life with different lubricants. The results for the lubricants used in these particular tests tend to indicate that thermal stresses due to temperature gradients in the contact zone of two rolling bodies may be of lesser importance in resolving differences in fatigue life than other factors discussed here. This is in contrast to the results obtained in reference 1. Other studies, however, strongly suggest that thermal effects merit certain considerations (ref. 25).

The contact temperature is a function of both the thermal and rheological properties of a lubricant. From an examination of the temperature data from table VII, summarized in figure 9, and an analysis of the variables that can affect the temperature, the rheological properties appear to be the more important. For the tests reported herein the contact temperature appears to be a function of viscosity, pressure-viscosity coefficient, and shear rate if the existence of a film in the contact area is assumed. Since the shear rate is a function of contact angle and since contact temperature varies linearly with contact angle for the five-ball fatigue tester (ref. 25), the slope is an index of the sensitivity of a lubricant to shear. Therefore, the slope of the lines in figure 9 indicates that the silicone and the mineral oils are more sensitive to shear than the esters. The effect of "relaxation" or "fluid memory" was not evaluated. However, this phenomenon can have a significant effect on the fluid viscosity (ref. 24) and thus contact temperature under dynamic conditions.

EVALUATING COMMENTS

While much is known about the operating characteristics of presently available commercial lubricants and their effect on rolling-contact fatigue, the application of this knowledge is still in the form of an art rather than a science. Attempts have been made to formulate an elastohydrodynamic theory in order to predict film thickness under dynamic conditions between rolling elements (refs. 1 and 26 to 32). These attempts are meeting with various degrees of success, but are presently limited by the lack of knowledge of both the Newtonian and non-Newtonian properties of lubricating fluids. However, should any theory for predicting film thickness be successful, an attempt must be made to relate it to the fatigue phenomenon. In that effort, both the effect of stress redistribution due to elastohydrodynamic lubrication and heat generation must be considered.

SUMMARY OF RESULTS

The NASA fatigue spin rig and five-ball fatigue tester were used to determine the rolling-contact fatigue properties of a group of lubricants of different base stock, viscosity, and additives. The following results were obtained:

1. A correlation is suggested between the plastically deformed profile radius (which was affected by the rheological properties of the lubricant) and rolling-contact fatigue life.

2. Fatigue life was not related to contact temperature based on data obtained with nine different lubricants.

3. Longer fatigue lives at room temperature were generally observed with both the chlorinated methyl-phenyl silicone and the di-2-ethylhexyl sebacates than with other lubricants. At 300° F, however, the silicone and the mineral oils gave the longer lives.

4. Tests run in the five-ball fatigue tester appear to be valid for rating the performance of different lubricants in rolling-contact fatigue. The relative order of the results in the five-ball fatigue tester compared favorably with that obtained for 7208-size bearings with the same lubricants.

5. Failure type and appearance were similar for balls run with all of the nine lubricants studied in the bench rigs.

Lewis Research Center
National Aeronautics and Space Administration
Cleveland, Ohio, June 29, 1962

APPENDIX A

SYMBOLS

a — deformation and wear area from surface trace, sq in.

B_{10} — ball-specimen 10-percent life, millions of stress cycles

B_{50} — ball-specimen 50-percent life, millions of stress cycles

C — thrust-load capacity, the load at which 90 percent of group of bearings can endure 1 million inner-race revolutions, or, for ball specimens, 1 million stress cycles, $(C = P \sqrt[3]{L})$, lb

C_B — ball-load capacity of specimens in NASA fatigue spin rig, lb

C_S — system thrust-load capacity of NASA five-ball fatigue tester, lb

C_T — thrust-load capacity of full-scale bearings, lb

H — depth of running track from surface trace, in.

h — maximum distance from the chord to the profile, in.

L — bearing or ball-specimen 10-percent life, or life at which 90 percent of a group of specimens remain unfailed, millions of stress cycles or inner-race revolutions

L_{10} — bearing 10-percent life, millions of inner-race revolutions

L_{50} — bearing 50-percent life, millions of inner-race revolutions

P — bearing or test-system thrust load, lb

R — radius of ball, in.

r — effective radius of ball profile after plastic deformation and wear, in.

x — one-half track width on ball specimen after deformation, in.

β — free-contact angle, deg

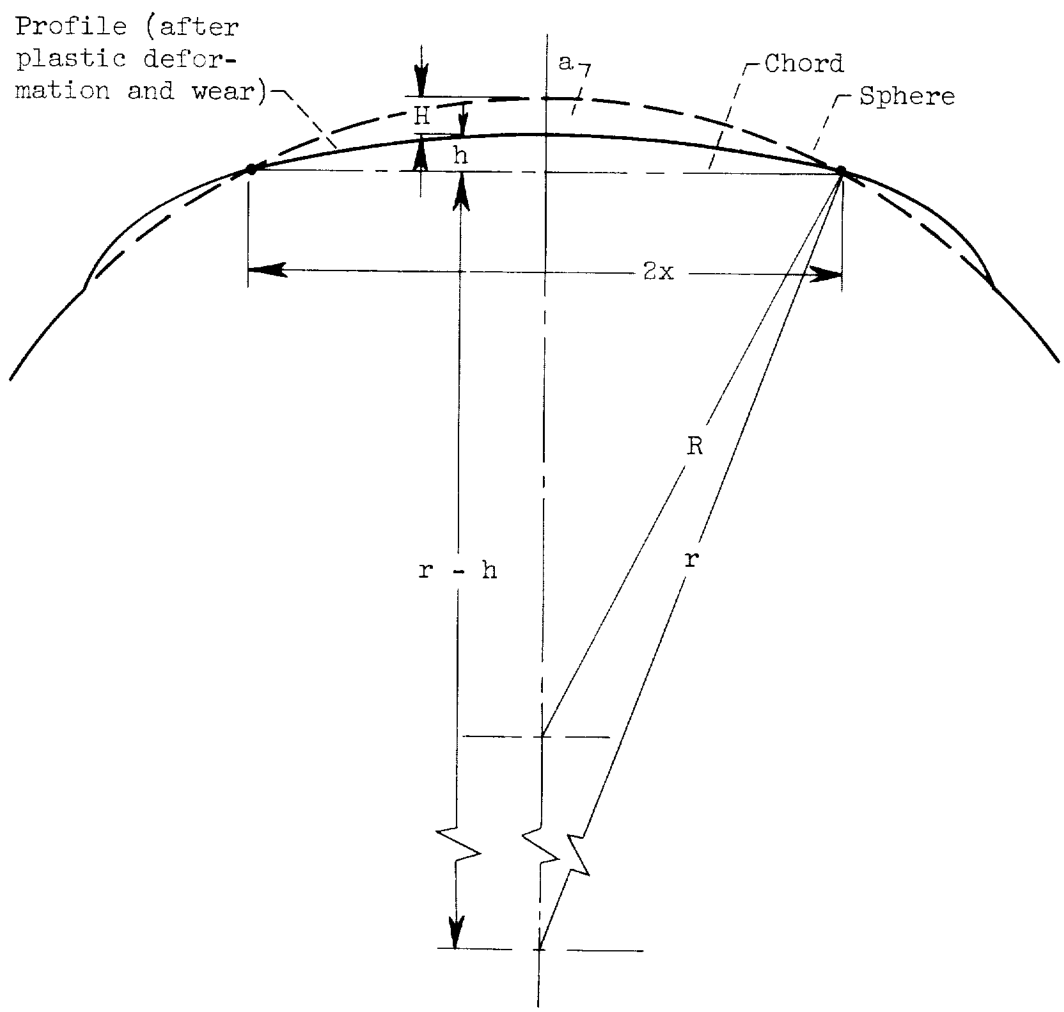
ω_i angular velocity of drive shaft, rpm

ω_s angular velocity of spin of ball specimen with respect to support balls, rpm

APPENDIX B

DERIVATION FOR EFFECTIVE-PROFILE RADIUS r

The geometry used in the derivation of the effective-profile radius after plastic deformation and wear is given in the following sketch:



Let $x = a/H$,

h = the maximum distance from the chord to the profile,

and

$$h' = h + H \qquad \text{or} \qquad h = h' - H \tag{B1}$$

Then

$$x^2 = r^2 - (r - h)^2$$

$$= 2rh - h^2 \qquad \text{(B2)}$$

and

$$r = \frac{x^2 + h^2}{2h} \qquad \text{(B3)}$$

If $r = R$ and $h = h'$, equation (B2) becomes

$$x^2 = 2Rh' - (h')^2$$

and

$$h' = R \pm \sqrt{R^2 - x^2} \qquad \text{(B4)}$$

Substituting equation (B4) into equation (B1) and solving for h in terms of a, R, X, and H result in:

$$h = R \pm \sqrt{R^2 - (a/H)^2} - H \qquad \text{(B5)}$$

If h is substituted into equation (B3) and a negative sign is selected for the radical in equation (B5),

$$r = \frac{(a/H)^2 + \left[R - \sqrt{R^2 - (a/H)^2} - H\right]^2}{2\left[R - \sqrt{R^2 - (a/H)^2} - H\right]} \qquad \text{(B6)}$$

Since $\left[R - \sqrt{R^2 - (a/H)^2} - H\right]^2$ is extremely small relative to $(a/H)^2$,

$$r = \frac{(a/H)^2}{2\left[R - \sqrt{R^2 - (a/H)^2} - H\right]}$$

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TABLE I. - TEST-LUBRICANT PROPERTIES[a]

Lubricant type	Lubricant designation	Base stock	Additive content		Neutralization number (before test)	Viscosity, cs (see fig. 2)		
			Type	Volume, percent		100° F	210° F	300° F
Ester	NA-XL-1	Pentaerythritol ester	(1) Oxidation inhibitor (2) Extreme-pressure additive (3) Antifoam agent	Unavailable	0.05	28.22	5.19	2.45
	NA-XL-2	Tri-methylol propane ester	(1) Oxidation inhibitor (1% pana + 1% 5-10-10)[a]	2.0	0.07	15.92	3.55	1.80
	NA-XL-3	Complex ester of octyl alcohol, adipic acid, and polyethylene glycol	(1) Oxidation inhibitor	0.5	0.18	35.30	7.14	3.40
	NA-XL-5	Di-2-ethylhexyl sebacate (dioctyl sebacate)	(1) Oxidation inhibitor (2) Extreme-pressure additive (3) Antifoam agent	0.5 1.0 .0005	0.25	13.24	3.40	1.83
	NA-XL-8	Di-2-ethylhexyl sebacate (dioctyl sebacate)	(1) Oxidation inhibitor (2) Extreme-pressure additive (3) Antifoam agent	0.5 5.0 .0005	0.25	13.29	3.51	1.86
Mineral oil	NA-XL-4	% Carbon paraffinic-63 % Carbon naphthenic-32 % Carbon aromatic-5	(1) Extreme-pressure additive	2.0	0.09	126.60	11.72	4.30
	NA-XL-6	% Carbon paraffinic-63 % Carbon naphthenic-35 % Carbon aromatic-2	(1) Antifoam agent	0.001	0.15	141.00	12.72	4.60
	NA-XL-7	% Carbon paraffinic-66 % Carbon naphthenic-33 % Carbon aromatic-1	No additives	------	0.10	238.70	20.68	7.00
Silicone	NA-XL-9	Chlorinated methylphenyl silicone	No additives	------	0.08	58.59	19.40	10.80

[a]See ref. 19 for additional properties.

TABLE II. - TEST-LUBRICANT FATIGUE RESULTS

Lubricant type	Lubricant designation	Fatigue spin rig; room temperature; maximum Hertz stress, 725,000 psi (a)			7205-size bearings; temperature, 300° F; thrust load, 4600 lb (b)			Five-ball fatigue tester; temperature, 300° F; maximum Hertz stress, 800,000 psi (c)		
		10-percent life, B_{10}, stress cycles	50-percent life, B_{50}, stress cycles	Ratio of B_{10} to B_{10} with NA-XL-7	10-percent life, L_{10}, inner-race revolutions	50-percent life, L_{50}, inner-race revolutions	Ratio of L_{10} to L_{10} with NA-XL-7	10-percent life, B_{10}, stress cycles	50-percent life, B_{50}, stress cycles	Ratio of B_{10} to B_{10} with NA-XL-7
Ester	NA-XL-1	27.0×10^{6}	290.0×10^{6}	0.15	-------	--------	----	--------	---------	----
	NA-XL-2	30.0	400.0	.17	-------	--------	----	--------	---------	----
	NA-XL-3	31.5	420.0	.18	1.1×10^{6}	4.4×10^{6}	0.33	13.7×10^{6}	65.0×10^{6}	0.31
	NA-XL-5	87.0	1700.0	.48	1.6	9.8	.49	--------	---------	----
	NA-XL-8	205.0	1400.0	1.14	1.4	8.4	.42	[d]14.9	[d]60.0	[d].34
Mineral oil	NA-XL-4	53.0×10^{6}	710.0×10^{6}	0.29	2.5×10^{6}	17.0×10^{6}	0.76	42.5×10^{6}	132.0×10^{6}	0.96
	NA-XL-6	180.0	1800.0	1.00	-------	--------	----	--------	---------	----
	NA-XL-7	180.0	1030.0	1.00	3.3	21.0	1.00	44.5	395.0	1.00
Silicone	NA-XL-9	500.0×10^{6}	4400.0×10^{6}	2.78	5.1×10^{6}	37.0×10^{6}	1.55	--------	---------	----

[a]Data plotted in fig. 4.
[b]Data plotted in fig. 5.
[c]Data plotted in fig. 6.
[d]From different lubricant batch.

TABLE III. - CONFIDENCE LEVELS OF FATIGUE DATA

Lubricant type	Lubricant designation	10-percent life confidence number, percent (compared to NA-XL-7)		
		Room temperature	300° F	
		Fatigue spin rig	7208-size bearings	Five-ball fatigue tester
Ester	NA-XL-1	90	----	----
	NA-XL-2	88	----	----
	NA-XL-3	87	87	78
	NA-XL-5	65	72	----
	NA-XL-8	53	86	81
Mineral oil	NA-XL-4	78	58	53
	NA-XL-6	50	----	----
	NA-XL-7	(a)	(a)	(a)
Silicone	NA-XL-9	70	62	----

[a]Reference lubricant.

TABLE IV. - LOAD CAPACITY OF TEST LUBRICANTS

Lubricant type	Lubricant designation	Fatigue spin rig; room temperature		7206-size bearings; 300° F		Five-ball fatigue tester; 300° F	
		Ball normal-load-carrying capacity, C_B, lb	Ratio of C_B to C_B with NA-XL-7	Bearing thrust-load-carrying capacity, C_T, lb	Ratio of C_T to C_T with NA-XL-7	Test-system thrust-load-carrying capacity, C_S, lb	Ratio of C_S to C_S with NA-XL-7
Ester	NA-XL-1	2340	0.53	----	----	----	----
	NA-XL-2	2420	.55	----	----	----	----
	NA-XL-3	2460	.56	4750	0.69	813	0.66
	NA-XL-5	3460	.78	5380	.79	----	----
	NA-XL-8	4600	1.04	5150	.75	842	.70
Mineral oil	NA-XL-4	2930	0.66	6240	0.91	1190	0.98
	NA-XL-6	4420	1.00	----	----	----	----
	NA-XL-7	4420	1.00	6850	1.00	1210	1.00
Silicone	NA-XL-9	6200	1.40	7910	1.15	----	----

TABLE V. - FATIGUE LIFE DUE TO FACTORS OTHER THAN LUBRICANT VISCOSITY

Lubricant type	Lubricant designation	Ratio of adjusted 10-percent life to 10-percent life with NA-XL-7		
		Room temperature; fatigue spin rig	300° F	
			7208-size bearings	Five-ball fatigue tester
		Adjusted to viscosity of 238.7 cs	Adjusted to viscosity of 7.0 cs	
Ester	NA-XL-1	0.26	----	----
	NA-XL-2	.33	----	----
	NA-XL-3	.29	0.40	0.37
	NA-XL-5	.99	.69	----
	NA-XL-8	2.35	.58	.47
Mineral oil	NA-XL-4	0.34	0.86	1.09
	NA-XL-6	1.14	----	----
	NA-XL-7	1.00	1.00	1.00
Silicone	NA-XL-9	3.95	1.40	----

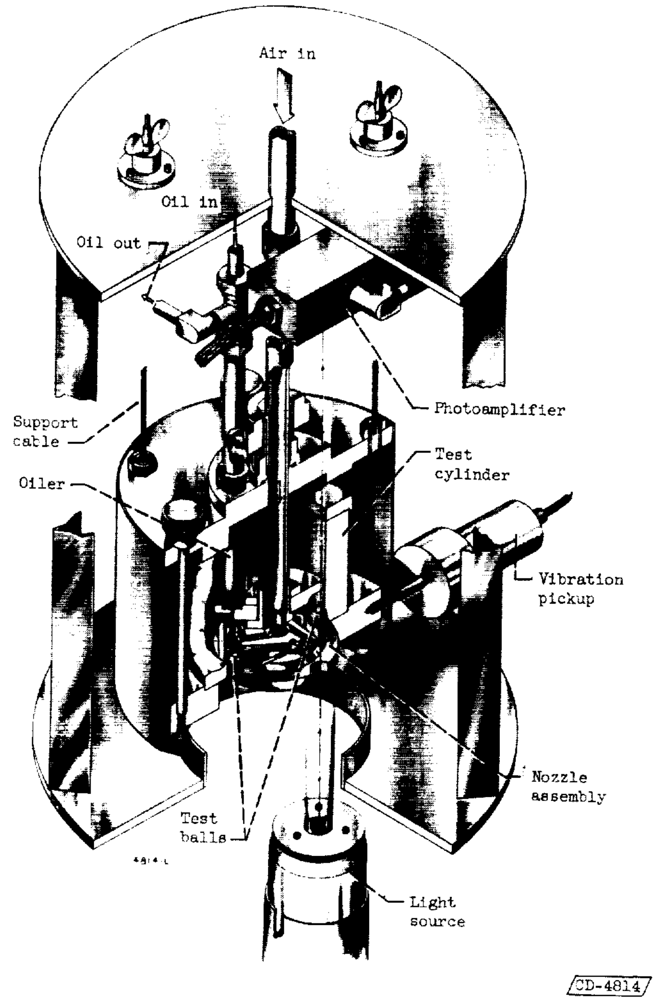
TABLE VI. - DEFORMATION AND WEAR AND THEIR EFFECT ON MAXIMUM HERTZ STRESS FOR M-50 STEEL BALLS [TEST IN FIVE-BALL FATIGUE TESTER; ROOM TEMPERATURE; INITIAL MAXIMUM HERTZ STRESS, 800,000 PSI; CONTACT ANGLE, 10°; 30,000 STRESS CYCLES]

Lubricant type	Lubricant designation	Deformation area from surface trace, sq in.	Wear area from surface trace, sq in.	Calculated profile radius, in.	Effective maximum Hertz stress, psi	
					No deformation and wear of support balls assumed	Deformation and wear of support balls assumed equal to test ball
Ester	NA-XL-1	3.66×10^{-7}	4.67×10^{-7}	0.320	7.68×10^{5}	7.36×10^{5}
	NA-XL-2	4.00	4.83	.321	7.67	7.35
	NA-XL-3	4.17	3.50	.329	7.65	7.30
	NA-XL-5	3.67	3.17	.329	7.64	7.29
	NA-XL-8	3.83	4.17	.322	7.67	7.34
Mineral oil	NA-XL-4	5.17×10^{-7}	4.33×10^{-7}	0.346	7.59×10^{5}	7.18×10^{5}
	NA-XL-6	4.83	4.33	.336	7.62	7.24
	NA-XL-7	4.33	4.00	.335	7.63	7.25
Silicone	NA-XL-9	8.83×10^{-7}	6.33×10^{-7}	0.340	7.61×10^{5}	7.22×10^{5}

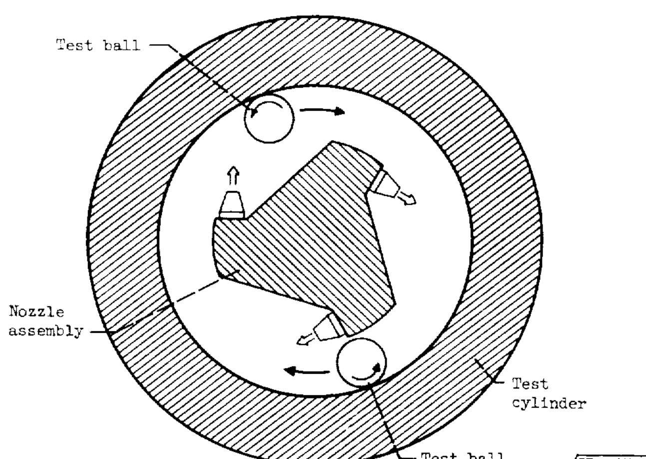
TABLE VII. - TEMPERATURE AT EDGE OF CONTACT ZONE [TEST IN MODIFIED FIVE-BALL FATIGUE TESTER; SPEED, 10,000 RPM]

Lubricant type	Lubricant designation	Temperature, °F (a)		
		725,000-psi maximum Hertz stress		800,000-psi maximum Hertz stress
		$\beta = 10^\circ$	$\beta = 30^\circ$	$\beta = 30^\circ$
Ester	NA-XL-1	121	142	---
	NA-XL-2	121	139	---
	NA-XL-3	117	141	161
	NA-XL-5	116	131	---
	NA-XL-8	118	133	156
Mineral oil	NA-XL-4	141	187	213
	NA-XL-6	127	180	---
	NA-XL-7	146	189	218
Silicone	NA-XL-9	154	196	---

[a]No heat added.

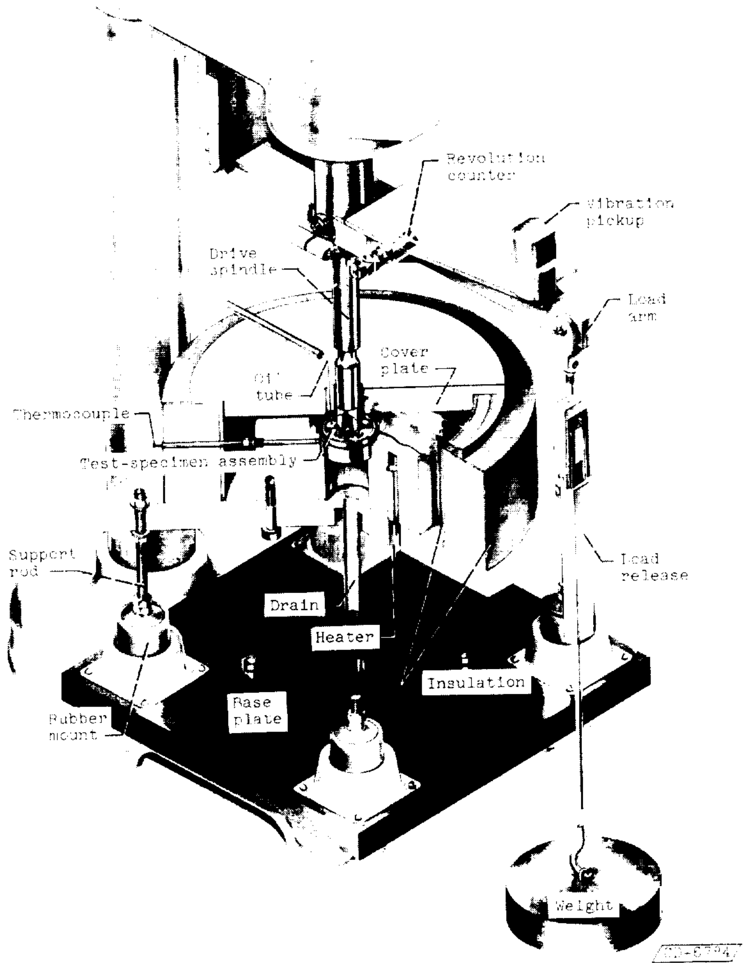


(a) Cutaway view of fatigue spin rig.

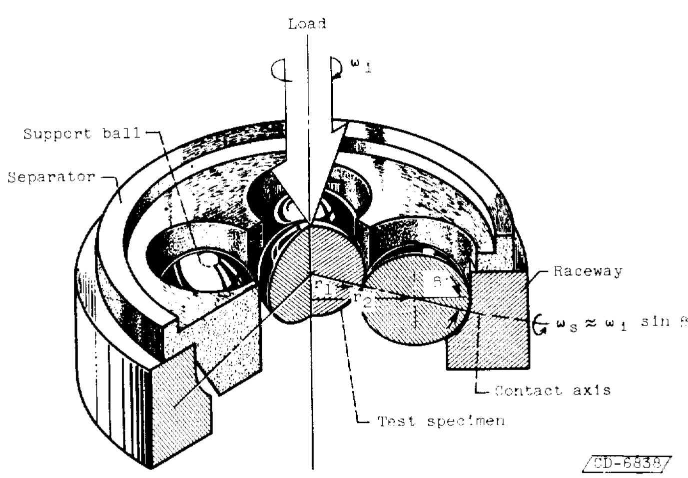


(b) Schematic of fatigue spin rig.

Figure 1. - Test apparatus.

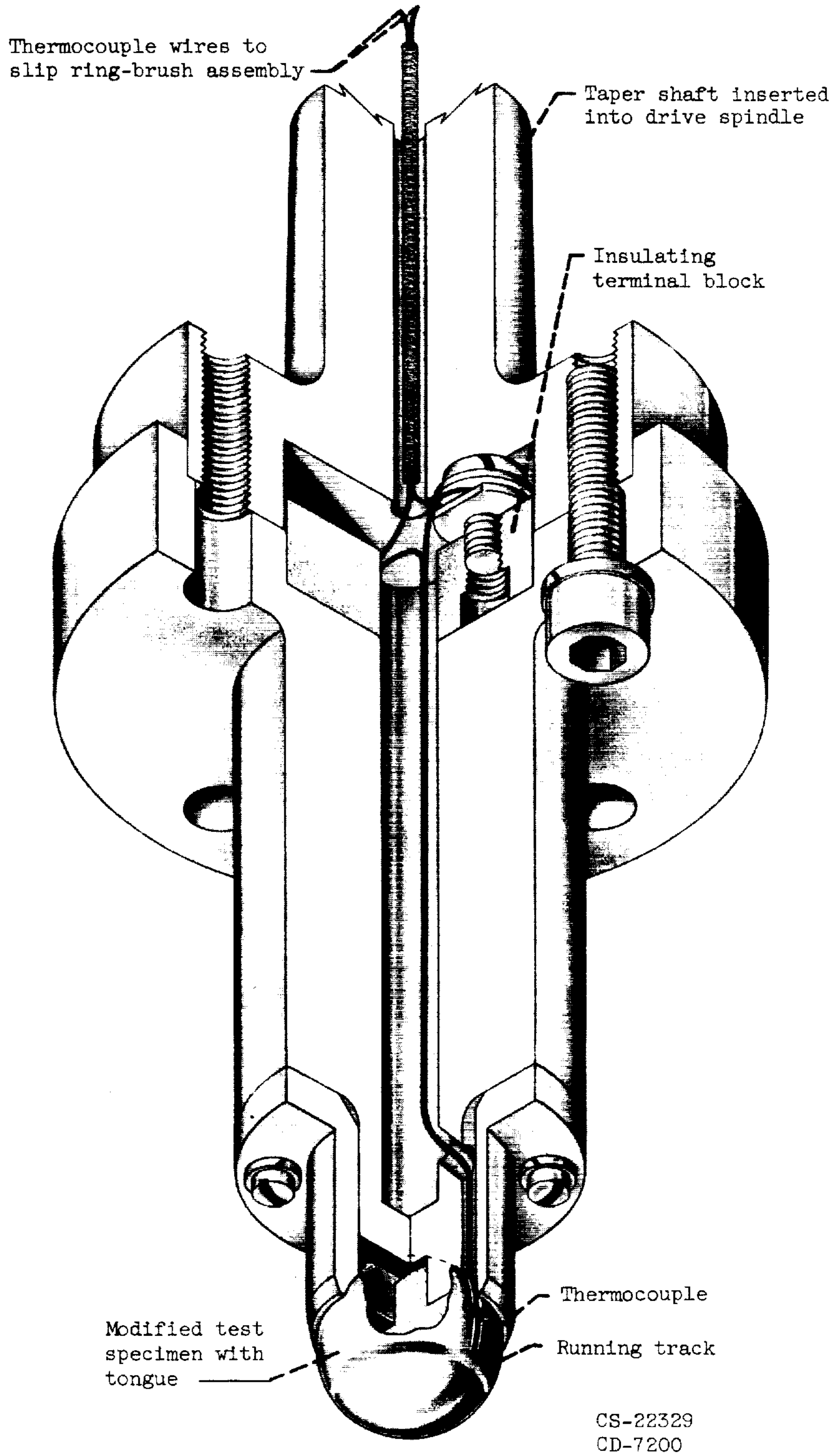


(c) Cutaway view of five-ball fatigue tester.



(d) Schematic of five-ball fatigue tester.

Figure 1. - Continued. Test apparatus.



(e) Operating-temperature measuring device.

Figure 1. - Concluded. Test apparatus.

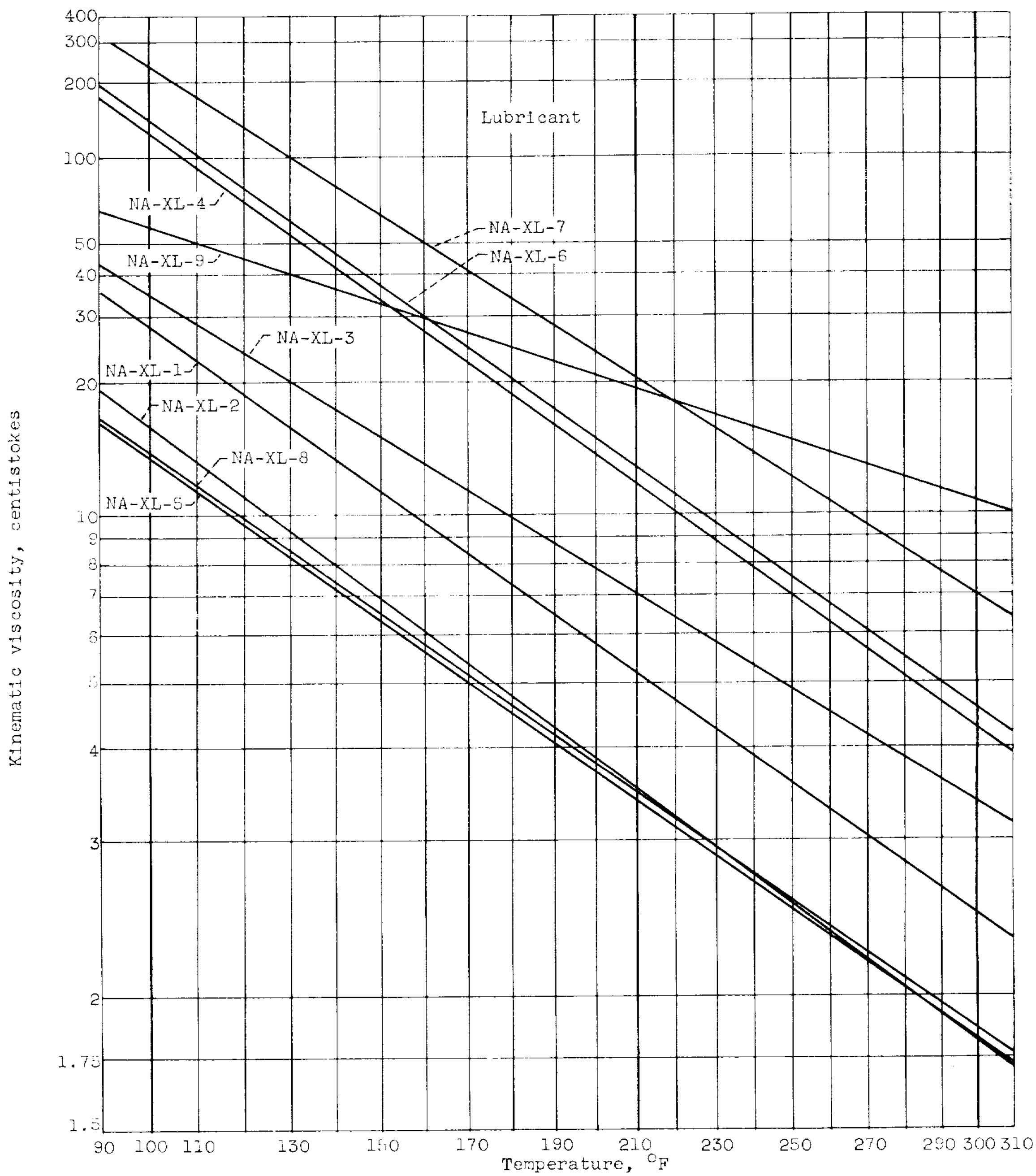
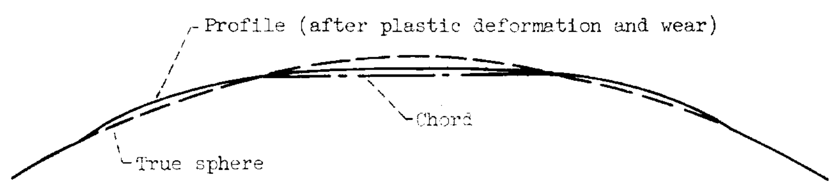
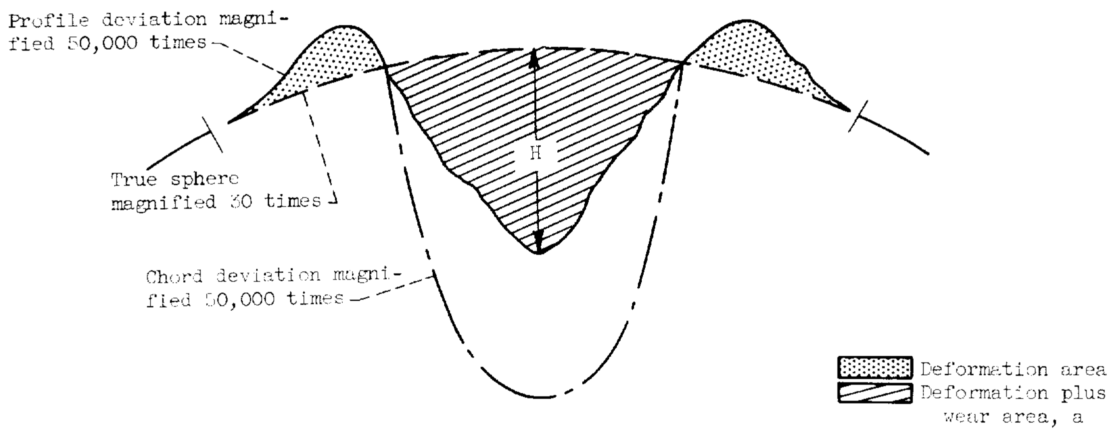


Figure 2. - Viscosity of experimental lubricants as function of temperature.

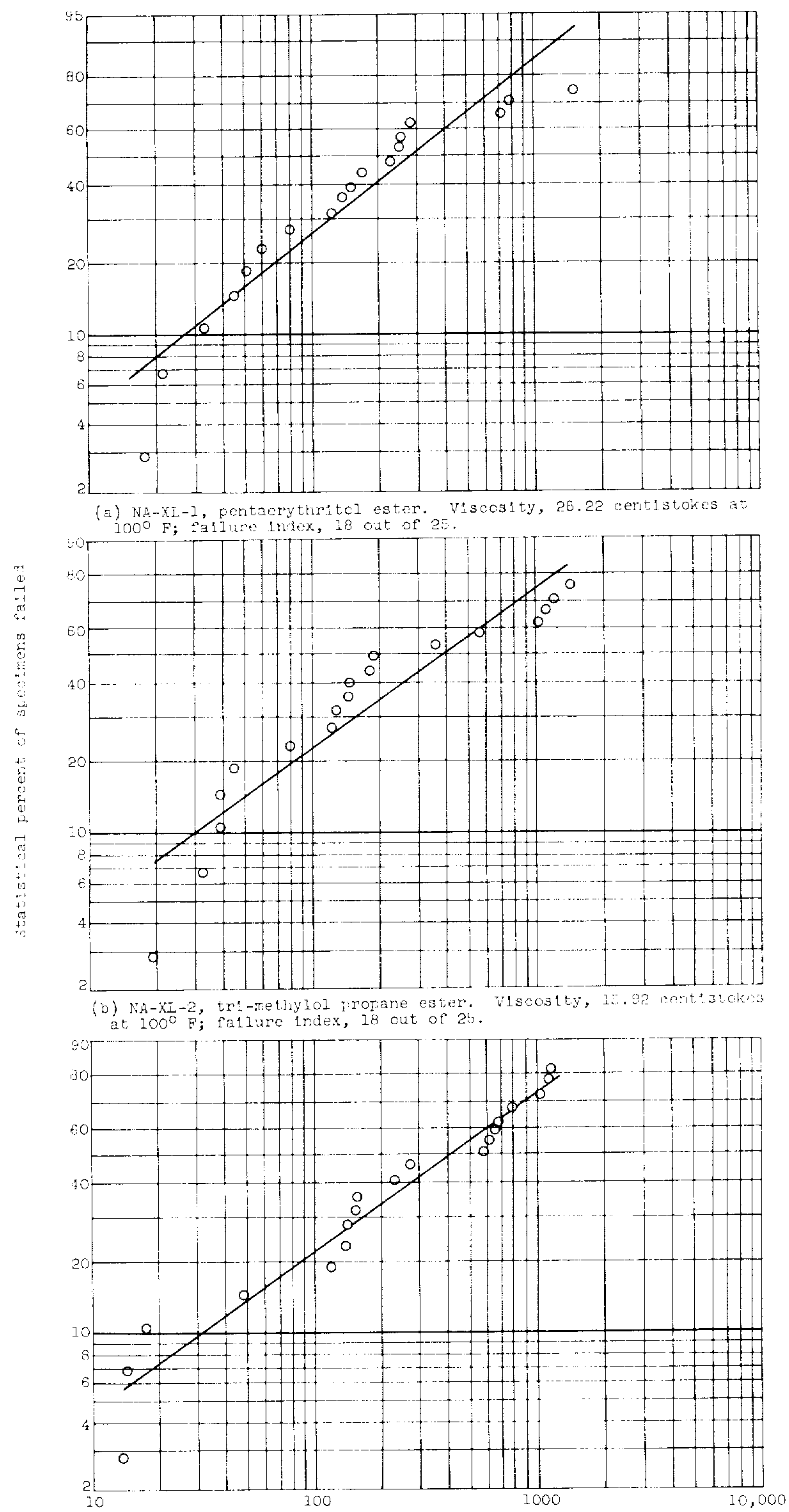


(a) Schematic diagram of transverse section of ball surface.



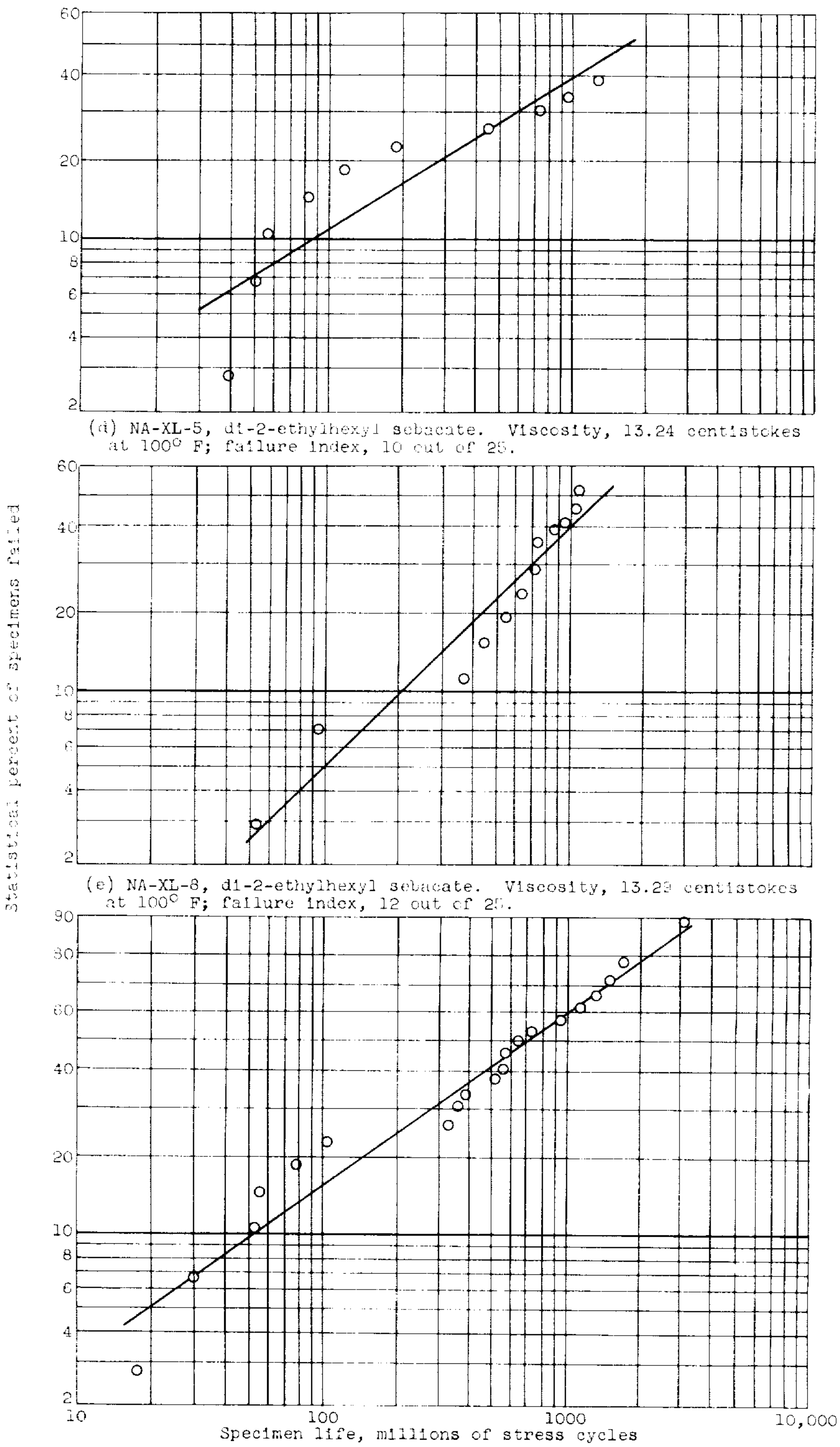
(b) Schematic diagram of surface trace of transverse section with deviation from true sphere highly magnified.

Figure 3. - Cross section of stressed ball track (not to scale).



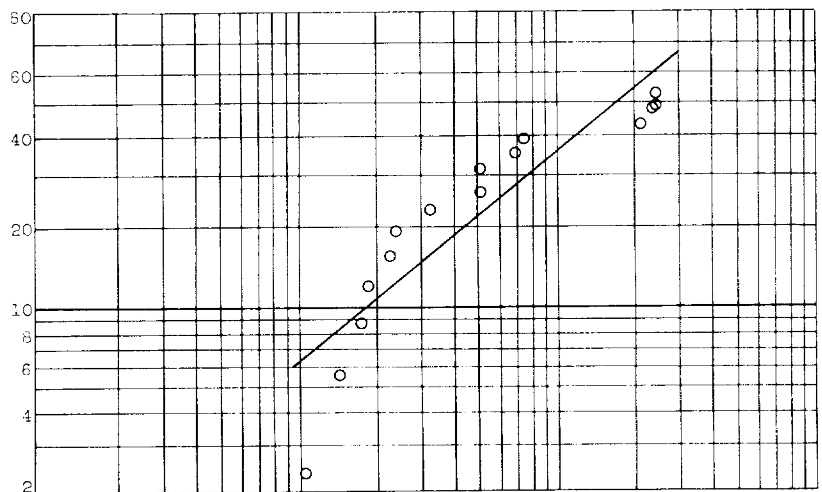
(c) NA-XL-3, complex ester. Viscosity, 35.3 centistokes at 100° F; failure index, 19 out of 25.

Figure 4. - Rolling-contact fatigue life of AISI M-2 steel balls with various lubricants. Test in fatigue spin rig; room temperature; maximum Hertz stress, 725,000 psi.

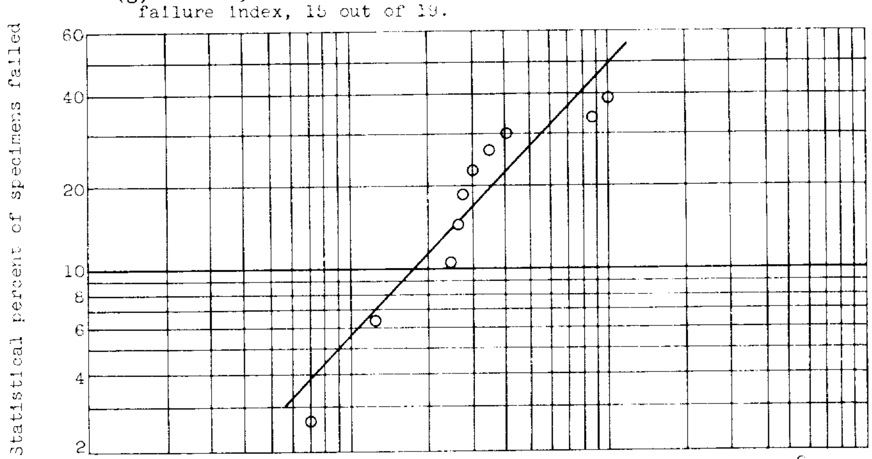


(f) NA-XL-4, mineral oil. Viscosity, 126.6 centistokes at 100° F; failure index, 20 out of 25.

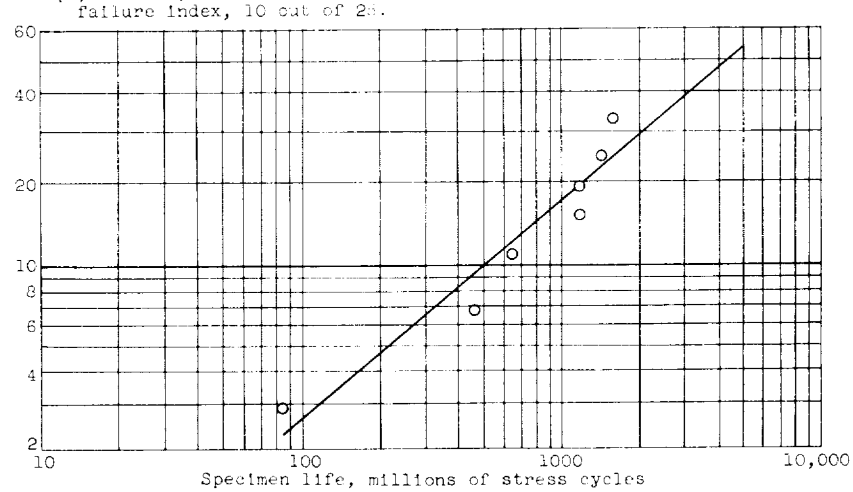
Figure 4. - Continued. Rolling-contact fatigue life of AISI M-2 steel balls with various lubricants. Test in fatigue spin rig; room temperature; maximum Hertz stress, 725,000 psi.



(g) NA-XL-6, mineral oil. Viscosity, 141.0 centistokes at 100° F; failure index, 15 out of 19.

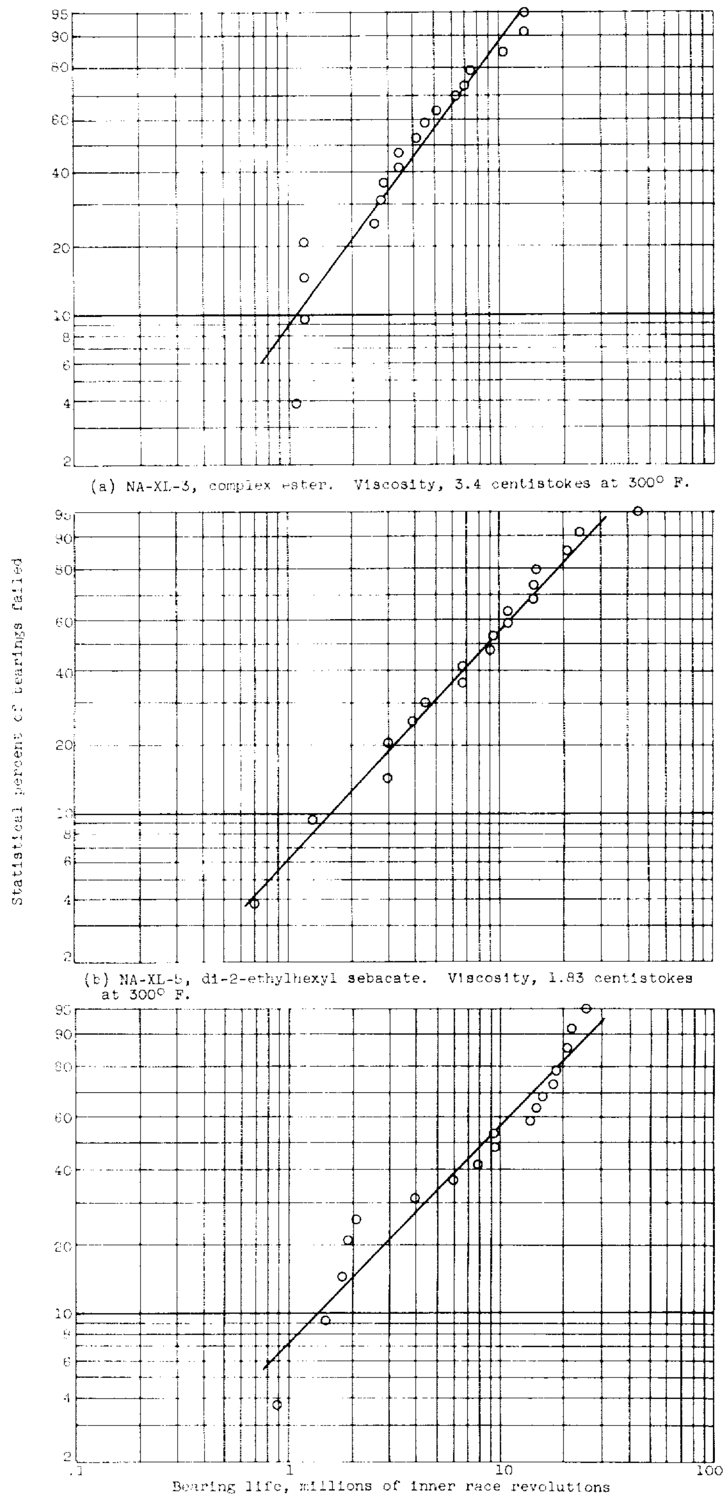


(h) NA-XL-7, mineral oil. Viscosity, 238.7 centistokes at 100° F; failure index, 10 out of 28.



(i) NA-XL-9, chlorinated methyl-phenyl silicone. Viscosity, 58.59 centistokes at 100° F; failure index, 7 out of 25.

Figure 4. - Concluded. Rolling-contact fatigue life of AISI M-2 steel balls with various lubricants. Test in fatigue spin rig; room temperature; maximum Hertz stress, 725,000 psi.

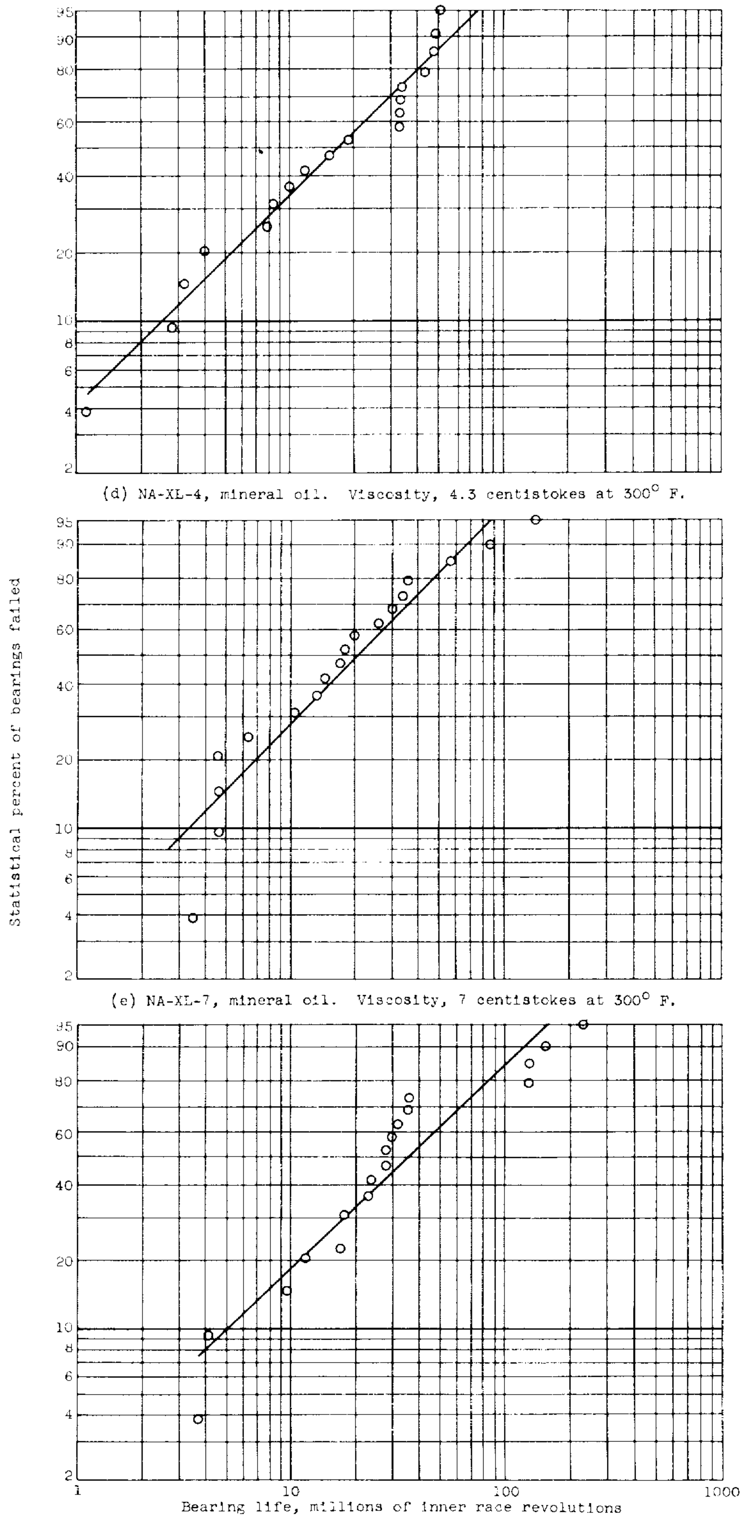


(a) NA-XL-3, complex ester. Viscosity, 3.4 centistokes at 300° F.

(b) NA-XL-5, di-2-ethylhexyl sebacate. Viscosity, 1.83 centistokes at 300° F.

(c) NA-XL-8, di-2-ethylhexyl sebacate. Viscosity, 1.86 centistokes at 300° F.

Figure 5. - Rolling-contact fatigue life of 7208-size ball bearings with various lubricants. Temperature, 300° F; speed 3450 rpm; axial load, 4600 pounds; failure index, 18 out of 18 (ref. 22).



(d) NA-XL-4, mineral oil. Viscosity, 4.3 centistokes at 300° F.

(e) NA-XL-7, mineral oil. Viscosity, 7 centistokes at 300° F.

(f) NA-XL-9, chlorinated methyl-phenyl silicone. Viscosity, 10.8 centistokes at 300° F.

Figure 5. - Concluded. Rolling-contact fatigue life of 7208-size ball bearings with various lubricants. Temperature, 300° F; speed, 3450 rpm; axial load, 4600 pounds; failure index, 18 out of 18 (ref. 22).

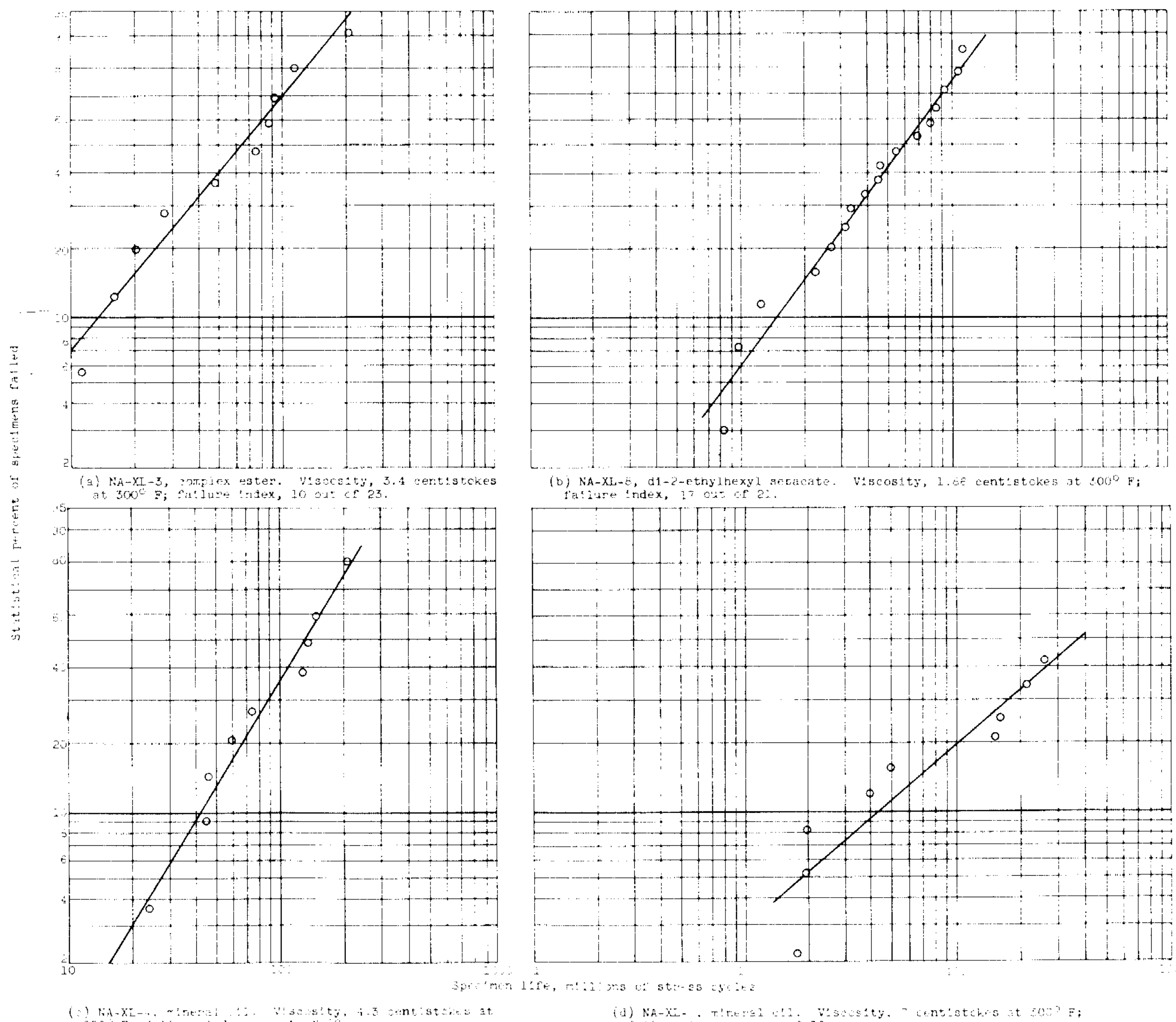


Figure 6. - Rolling-contact fatigue life of AISI M-1 steel balls with various lubricants. Test in five-ball fatigue tester; temperature, 300° F; maximum Hertz stress, 800,000 psi; contact angle, 30°.

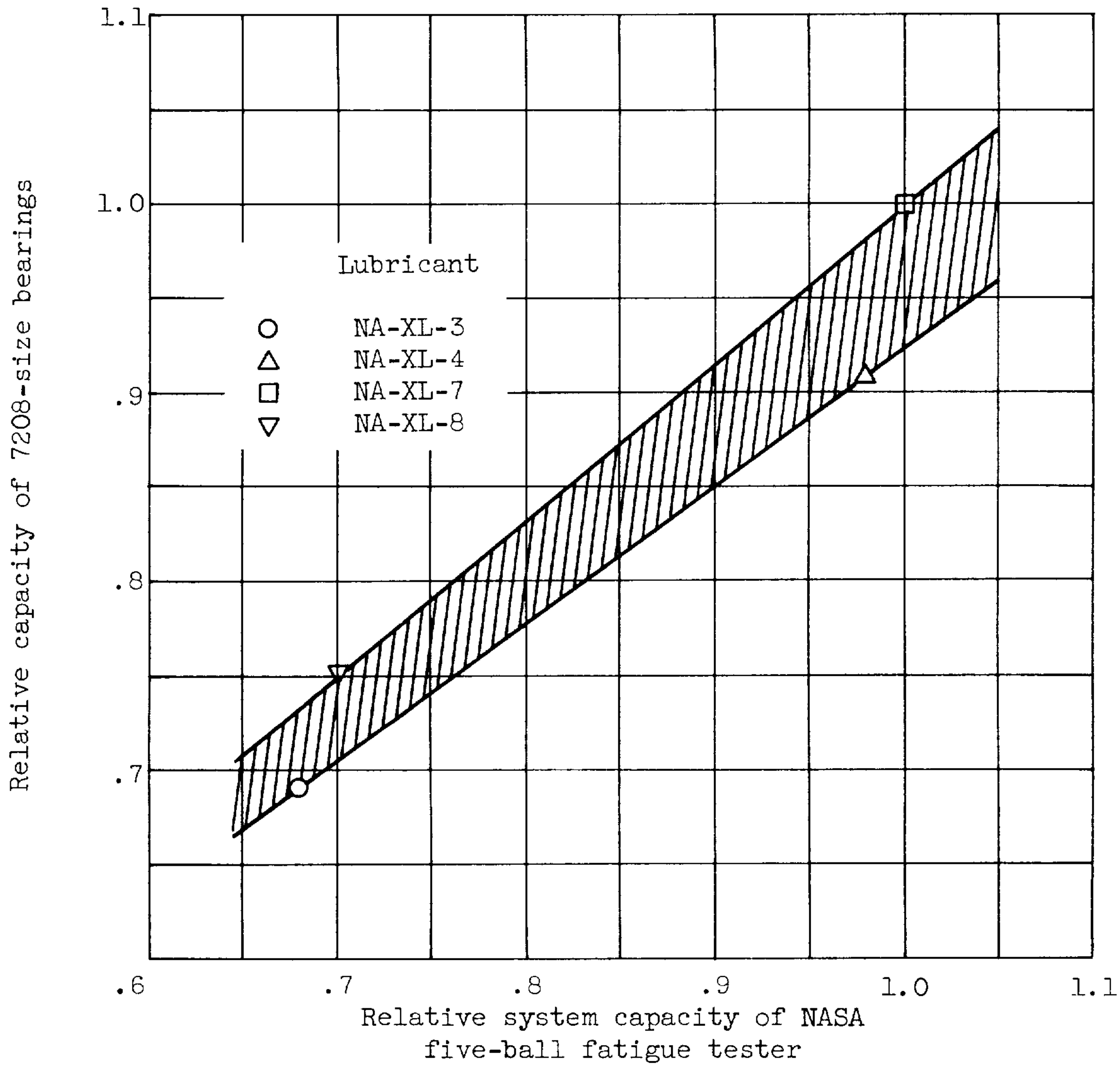


Figure 7. - Relative load-carrying capacity of 7208-size bearings as function of relative capacity of five-ball fatigue tester for various lubricants at 300° F.

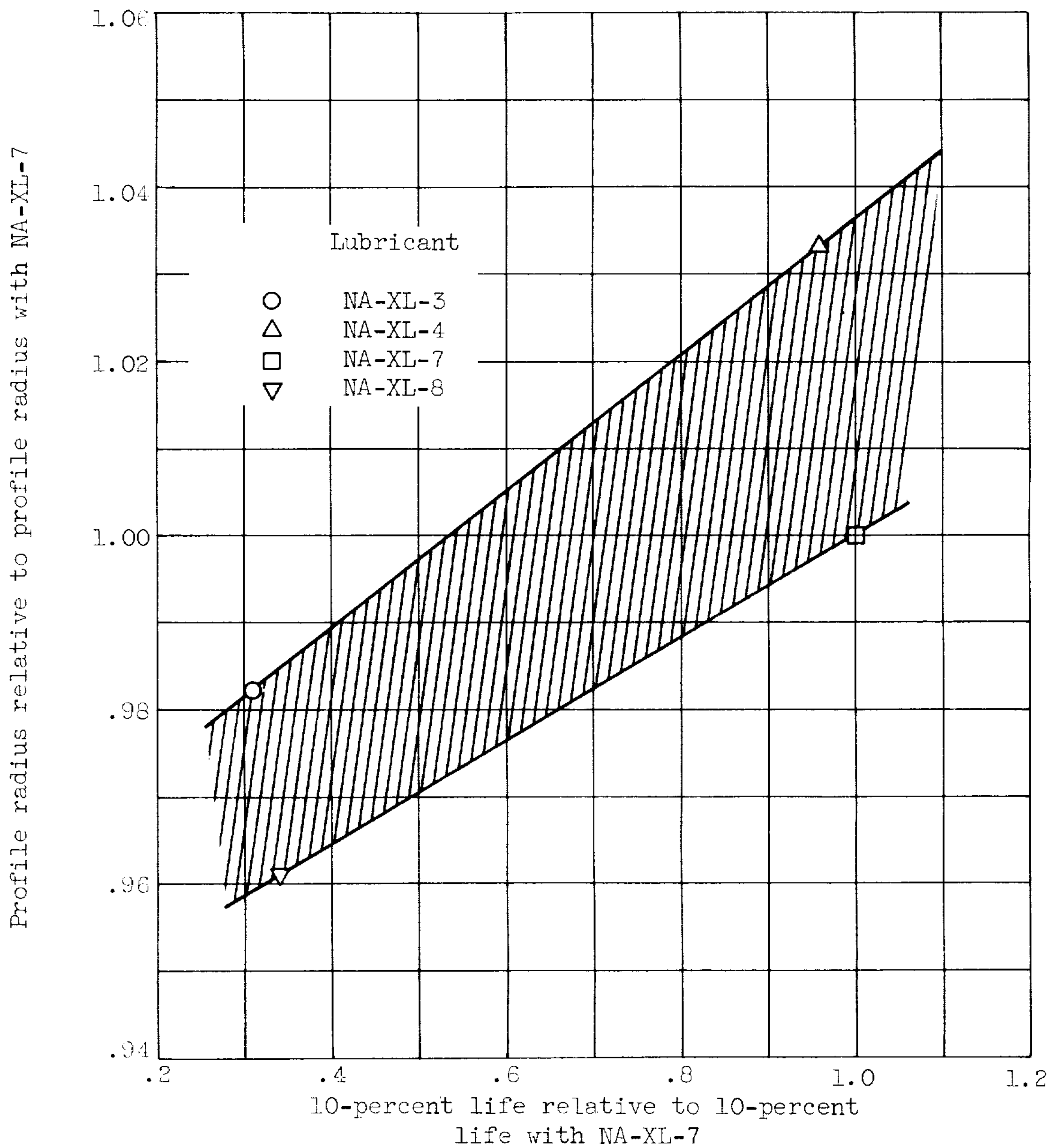


Figure 8. - Ball profile radius measured at 10° contact angle with no heat added as function of 10-percent life at 30° contact angle with outer-race temperature of 300° F. Test in five-ball fatigue tester; initial maximum Hertz stress, 800,000 psi.

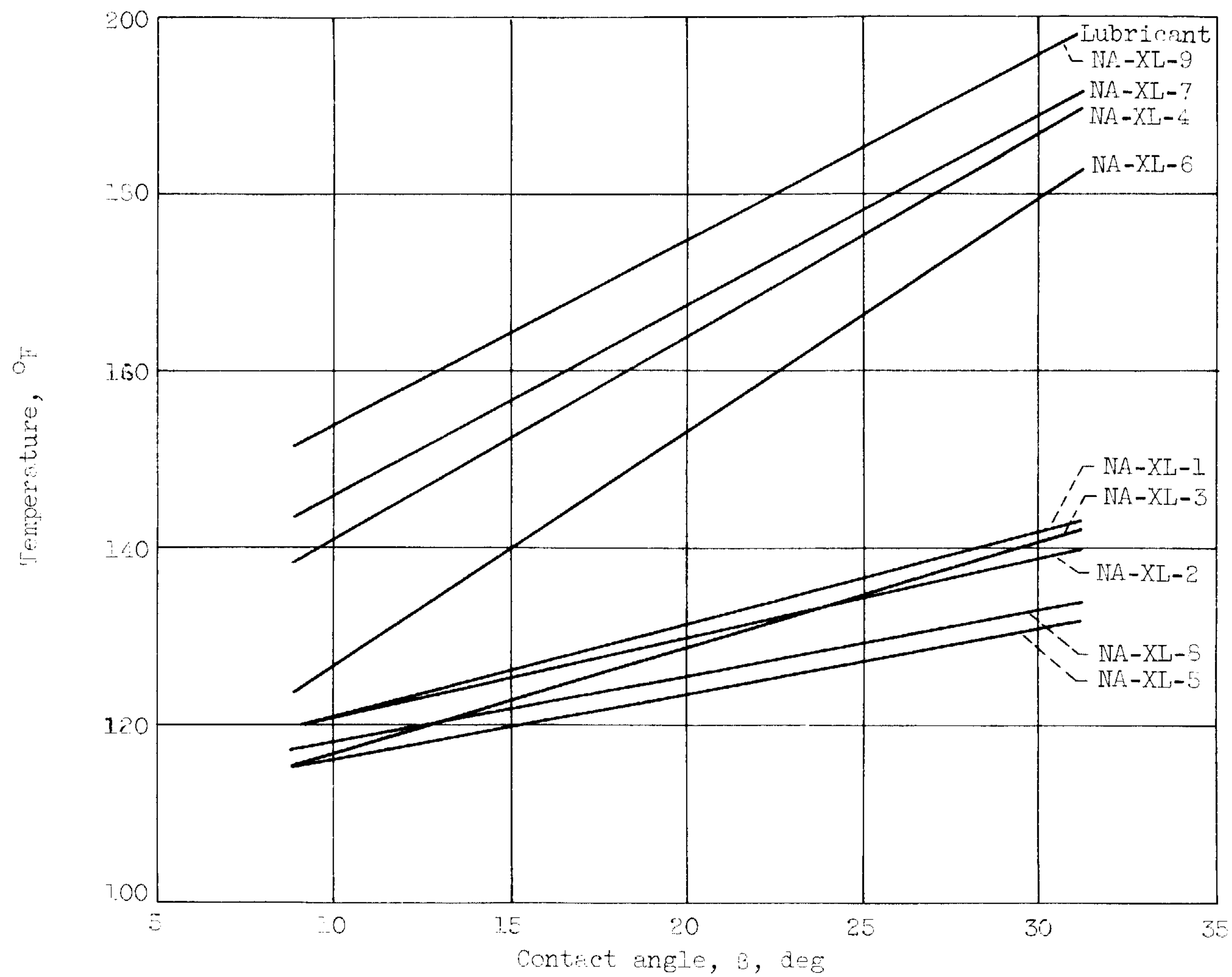


Figure 9. - Temperature at edge of contact zone as function of contact angle. Test in five-ball fatigue tester; no heat added; speed, 10,000 rpm; maximum Hertz stress, 725,000 psi. Data from table VII.